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THE SMALL DIESEL ENGINE AS AN AUTOMOTIVE ENGINE

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The scarcity and high cost of the light hydrocarbons has led to a great variety of experiments for utilizing heavier fuels, like petroleum, crude oil, and even tar oil, for driving automotive engines without making any important changes in the engines themselves. It was to be foreseen that this kind of experimenting would bring no satisfactory result. Aside from the excessive fuel consumption, which was generally over 300 grams per HP-hour, the main fault consisted in the rapid fouling of the engine from incomplete combustion

Since, up to the present time, the complete combustion of the heavier hydrocarbons has only been possible in the Diesel engine (aside from engines with much higher compression and hot-bulb engines, which can hardly be considered for motor-cars), nothing seems more natural than to fit out motor-cars with small Diesel engines Still, there is today no serviceable small Diesel engine available. There is, therefore, a wide-spread opinion that the Diesel engine in small sizes is not feasible, on account of a series of insurmountable difficulties. It should be remembered, however, that the reason for there being no practical small Diesel engine available is that no large firm with sufficient experience in building Diesel engines has, until recently, given the matter serious attention. These firms not only lack work-shops for producing motor

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car engines, but often also any understanding of the coming import-
ance of light engines. On the other hand, automobile firms lack
the valuable experience in building Diesel engines. In this re-
spect, and partly on account of the war, there have been many
changes, so that in the opinion of the author, within two years at
the latest the small Diesel engine will gradually replace the hith-
erto common automobile engine. It is assumed that the future en-
gine will be fully adapted to the new purpose, which will require
various structural changes in the regular Diesel engine. The main
characteristic of the Diesel engine, that of introducing the fuel
in the most finely divided condition possible into the highly heat-
ed air charge, must however be retained, since it is to this very
characteristic that the complete combustion is to be attributed.

Whether the introduction and spraying of the fuel is accom-
plished, as customary, by means of compressed air, or, as has rec-
ently been successfully done, by purely mechanical means (by the
pressure from a pump) is indifferent. Either method is adapted
to the small Diesel engine in question, as has been demonstrated
in experiments by the author on an ordinary engine and also on a
compressorless engine (by mechanical spraying). Which method is to
be used must be determined for each case.

Warning must be given against "semi-Diesel" engines and the
"similar-to-Diesel" engines. Engineswith incomplete cycles are im-
practicable as automotive engines, for which the best is none too
good.

For obvious reasons, not all the structural details of both ex-

perimental engines can be given here. We will give, however, the essential principles for building small Diesel engines, and also structural details, so far as feasible.

1. Crankcase.- The pressure in a Diesel engine is about one-third greater than in an explosion engine. (For strength calculations, 40 atm. are taken.) A complete Diesel engine is therefore 15 to 20% heavier than a regular automotive engine. Hence weight must be saved, especially in the crankcase. This renders it necessary for the upper part of the crankcase and the cylinder block to be cast together (of aluminum or cast-iron). This increases the strength in proportion to the smaller weight and at the same time reduces the cost of production. For the cylinders it is advantageous to insert steel liners. In the one case, the danger of wearing out is thereby diminished and in the other case, as a result of the better cooling, the efficiency is greatly increased; that is to say, assuming the balance of heat to remain the same, the efficiency can be increased (by increasing the r.p.m. and, so far as practicable, the fuel charge) in proportion to the efficacy of the cooling.

2. Crankshaft.- For raising the rather low mechanical efficiency in Diesel engines, it is desirable to place the crankshaft on roller or ball bearings. Since this kind of bearing is made narrow, we then obtain at the same time broad connecting rod bearings corresponding to the high pressures. For the sake of security there is a bearing on each side of every crank (for example, five bearings for a four-cylinder engine). The customary pressure lubri-

cation of the driving mechanism must, nevertheless, be retained.

3. <u>Pistons</u>.- Each piston has a deep hollow in the middle, so that the air for the combustion will be compressed under the fuel intake valve, as near the middle as possible. For the high compression here required, the piston then has in its upper dead center, on its outer circumference, only a small amount of play (about 2 min.) opposite the cylinder cover.

4. <u>Compression</u>.- The exponent of the compression line climbs, indeed, with the r.p.m. , but on the other hand, for the small size of the engine, the heat-radiating surface is large in proportion to the cylinder capacity. The best values are:

$$
\begin{aligned}
&\text{For } n = 800 \text{ to } 1000, \; \epsilon = 18; \\
&\text{For } n = 1000 \text{ to } 1400, \; \epsilon = 20.
\end{aligned}
$$

5. <u>Valve timing</u>.- Example (four-stroke cycle).

Fuel intake valve is open from 5° before top dead center to 40° after top dead center.

Air intake valve is open from 5° after top dead center to 40° after bottom dead center.

Exhaust valve is open from 45° before bottom dead center to 10° after top dead center.

Starting valve is open from 2° before top dead center to 120° after top dead center.

6. <u>Compressor for engines using compressed air</u>.- A compressor (with two or three stages) must be built with especial care. For the sake of simplicity, we will choose one with two stages. On account of the high condensation (about ninefold within one stage) the installation of an intermediate cooler is absolutely necessary. It is best to install the cooler concentrically around the compression cylinder. At the bottom of the cooler, an outlet cock or valve

must be provided for drawing off the oil and water from time to time.

In regard to the size and the choice of the stage-ratio, we will illustrate by means of an example: a two-stage truck-engine of 40-50 HP, 4 cylinder, 125 Φ x 150 stroke, n = 800-1000, Compressor drive at flywheel end of engine, with fifth crank. According to experience, the volume of the part of the compressor cylinder swept by its piston (= V_C) must be about 1/8 to 1/7 of the part of the working cylinder (= V_A) swept by its piston. The compressor can then sufficiently complete the reserve supply of air in the flasks.

$$V_A = \frac{1}{2} \; 4 \times 1.25^2 \; \frac{\pi}{4} \times 1.5 = 3.675 \text{ liters per revolution.}$$

$$V_C = \frac{3.675}{7} \div \frac{3.675}{8} = 0.525 \div 0.46 \quad " \qquad " \qquad "$$

In order not to obtain too small cylinder diameters, the stroke is taken at 100 mm. The compression cylinder does not then attain, even with a long piston rod, the height of the working cylinder.

At 90 mm. Φ of the normal-pressure stage and at 30 mm. Φ of the high-pressure stage $V_C = 0.565$ liter.

The receiver pressure would hereby be, under supposition of equal clearance volumes in both stages, 8 atmospheres, but only so long as the low pressure stage is exerting full suction, that is, only during the filling of the air flasks. If, on the contrary, during the normal drive, the air intake is throttled, this is then, in its effect, equivalent to a reduction of the low-pressure stage. The receiver pressure will therefore, with the above given diameters of both stages, fall about 6 atm. in its normal operation, that is, the normal pressure will work with only sixfold condensa-

tion and the high pressure, on the contrary, with about twelve to thirteenfold. In order to obtain nearly equal condensation in both stages, it would accordingly be desirable to make the high pressure stage still smaller than 30 mm. Φ . This is, however, not possible in practice, since already a piston diameter of 30 mm. creates difficulties in preventing leakage resulting from the extraordinarily high pressure. Instead of this, it is advisable to work with a purposely large clearance volume in the high pressure stage and to increase the diameter of the high pressure cylinder correspondingly. In proportion to the effect, the high pressure stage will then be smaller, without any resulting disadvantage from this expedient. For the above case, the high pressure cylinder can be conveniently made with 40 mm. Φ . V_C then becomes $(0.9^2 \frac{\pi}{4} - 0.4^2 \frac{\pi}{4}) \times 1 = 0.513$ liter per revolution, that is, it is large enough.

It is still important to determine the clearance volume of the high pressure stage which may here be carried out for the above example.

(Normal pressure) ND 90 Φ/40 Φ)
(High   ") ND 40 Φ) 100 stroke

If, for example, the normal pressure stage has 5% clearance volume and the receiver pressure is to be 9 atmospheres (with unthrottled intake), then the volumetric efficiency of the normal pressure, without taking account of the losses and heating of the intake air, is

$$\eta \text{ vol. ND} = 100\% - 5 (9 - 1) = 60\%$$

the isothermic course of the re-expansion line being taken for granted.

The actual air intake of the normal pressure stage then becomes

$$V_{ND} \text{ actual} = 513 \text{ cm.}^3 \times 0.6 = 307.8 \text{ cm.}^3 \text{ actual.}$$

For the receiver pressure to be 9 atmospheres,

V_{HD} actual $= \dfrac{307.8}{9} = 34.2$ cm.^{3}, or, since the part of the cylinder swept by the piston of the HD (high pressure) stage is 125 cm.^{3}, it becomes

$$\eta \text{ volume}_{HD} = \frac{34.2}{125} \times 100 = 27.4\%.$$

This requires a clearance volume (for the high pressure stage) of

$$V_{S_{HD}} = \frac{100 - 27.4}{9 - 1} = 9.8\%,$$

in case the high pressure stage is to produce a ninefold condensation.

If the polytropic re-expansion is assumed, for instance n = 413, then the clearance volume of the high pressure stage $V_{S_{HD}}$ is found by the following calculation:

$$\eta \text{ volume}_{ND} = 100\% - (9^{3/4} - 1) \times 5 = 79.02\%$$

$$V_{ND_{actual}} = 513 \text{ cm.}^3 \times 0.79 = 406 \text{ cm.}^3 \text{ actual.}$$

$$V_{HD_{actual}} = 406 \div 9 = 45.1 \text{ cm.}^3, \text{ or}$$

$$\eta \text{ volume}_{HD} = 45.1 \div 125 \times 100 = 36.08\%$$

Hence: $\underline{V_{S_{HD}}} = \dfrac{100 - 36.08}{9^{3/4} - 1} = \underline{15.23\%}$

The assumption n = 4/3 for the re-expansion line agrees approximately with the result of the experiment carried out with air pumps and may be taken as the basis for the determination of the

clearance volume. It is naturally within one's power to adjust the clearance volume in testing the machine (best by means of a screw cap closing the high pressure cylinder, which may at the same time contain the high pressure valve). It is even possible to adjust the clearance volume in the working of the intake air and also the end pressure, so as to obtain any desired pressure in the receiver. For the adjustment of the intake pressure the amount of the intake air is regulated by a hand throttle (or automatically). Herewith there may be simultaneously regulated the amount of air drawn in from the high pressure and also the pressure in the receiver.

In the foregoing it was shown that an intentional enlargement of the clearance volume in the high pressure stage may have its advantages under some circumstances. It will now be shown that, aside from the cylinder enlargement desired in the above instance, the clearance volume of the high pressure stage and, in general, the clearance volume of an air pump produces no disadvantage, since the size of the clearance volume has no influence on the energy required for condensing a given amount of air. (According to Hütte XX, Part I, p.339, the contrary might be expected, but in reality the clearance volume exerts no influence on the work requirement.) One may satisfy himself in regard to this in a simple manner, by drawing two diagrams for clearance volumes of different sizes and finding the relation of the diagram surface to the actual amount of air sucked in. As will be shown, the same value is obtained for this ratio in both cases.

In Fig. 1, diagram 1 2 3 4 1 corresponds to the clearance vol-

ume $V_S = 0$ and diagram I II III IV I to the clearance volume $V_S = 0.2$ V stroke. The condensation lines 1 and 2, as also I and II, are drawn as isotherms, as likewise the re-expansion lines III and IV. The "stroke-volume" V_H is assumed to be 10 cm., the initial pressure 1 cm. and the end pressure 2.5 cm.

1. The energy required without the clearance volume is equal to the surface 1 2 3 4 1 = surface 1 2 2' 1' = $A = \int_{V_2}^{V_1} p \, dv$. With $p \times v = C$

$$A = C \int_{V_2}^{V_1} \frac{dv}{v} = C \times \ln \frac{V_1}{V_2} = 10 \times 1 \times \ln 2.5 = 9.163 \text{ cm.}^2$$

The intake amount V_{sg} equals the distance $4 \div 1$, d.h. = 10 cm., and the ratio $\frac{A}{V_{sg}} = \frac{9.163}{10} = \underline{0.9163}$.

2. The energy required with the clearance volume $v_s = 20\%$ is equal to the surface I II III IV I = $A' = A_1 + A_2 - A_3 - A_4$.

$$\text{Surface} \quad \text{I II II' I'} = A_1$$
$$+ \quad '' \quad \text{II III III' II'} = + A_2$$
$$- \quad '' \quad \text{III IV IV' III'} = - A_3$$
$$- \quad '' \quad \text{IV I I' IV'} = - A_4$$

$A_1 = \int_{V_2}^{V_1} p \, dv = 12 \ln \frac{v_I}{v_{II}} = 12 \times \ln 2.5 = 10.9956 \text{ cm.}^2$

$A_2 = 2.5 \times 2.8 = 7 \text{ cm.}^2$

$A_3 = \int_{V_{III}}^{V_{IV}} p \, dv = C \times \ln 2.5 = 5 \times \ln 2.5 = 4.58145 \text{ cm.}^2$

$A_4 = 1 \times 7 = 7 \text{ cm.}^2 = A_2$

$A' = A_1 - A_3 = 10.9956 - 4.58145 = 6.41415 \text{ cm.}^2$

V_{sg} becomes $0.7 \, v_H = 7$ cm. and hence

$$\frac{A'}{V_{sg}} = \frac{6.41415}{7} = \underline{0.9163}, \quad \text{just the same as in Case 1.}$$

Without a numerical example, the proof may be carried out as follows, the designations being the same as above.

1) $v_S = 0$

$$p \times v = C = p_1 \, v_1 = p_2 \, v_2 .$$

$A = A_1 = $ absolute work of condensation

$$A = \int_{v_2}^{v_1} p\,dv = p_1 \, v_1 \int_{v_2}^{v_1} \frac{dv}{v} = p_1 \, v_1 \, \ln \frac{v_1}{v_2}$$

$$V_{sg} = v_1, \text{ hence}$$

$$\frac{A}{V_{sg}} \text{ for } v_S = 0 \quad = \frac{p_1 \, v_1 \, \ln \frac{v_1}{v_2}}{v_1} = \quad p_1 \, \ln \frac{p_2}{p_1}$$

2) $v_S = \alpha \times v \text{ stroke} = \alpha \times v_1.$

$$p \times v = C = p_1 \, v_1 = p_2 \, v_{II}$$

$$A_1 = \int_{v_{II}}^{v_I} p\,dv = p_I \, v_I \, \ln \frac{v_I}{v_{II}} = p_I \, v_I \, \ln \frac{v_1}{v_2} =$$

$$(1 + \alpha) \, p_1 \, v_1 \, \ln \frac{v_1}{v_2}$$

$A_2 = $ absolute exhaust work $= - A_4 = $ absolute intake work.

$A_3 = $ re-expansion work $= \int_{v_{III}}^{v_{IV}} p\,dv =$

$$p_{III} \, v_{III} \, \ln \frac{v_{IV}}{v_{III}} = p_3 \, v_6 \, \ln \frac{p_2}{p_1} = p_2 \, v_S \, \ln \frac{v_1}{v_2} =$$

$$\alpha \times v_1 \times p_2 \times \ln \frac{v_1}{v_2}$$

Hence $A' = A_1 - A_3 = (1 + \alpha) \, p_1 \, v_1 \, \ln \frac{v_1}{v_2} -$

$$\alpha \times v_1 \times p_2 \times \ln \frac{v_1}{v_2}$$

Since $V_{sg} = v_I - v_{IV} = v_1 + \alpha v_1 - v_s \dfrac{p_2}{p_1} =$

$$(1 + \alpha) v_1 - \alpha v_1 \dfrac{p_2}{p_1},$$

Therefore $\dfrac{A'}{V_{sg}}$ for $v_s = \alpha \times 100\%$ $=$

$$\frac{(1 + \alpha) p_1 v_1 \ln \dfrac{v_1}{v_2} - \alpha v_1 p_2 \ln \dfrac{v_1}{v_2}}{(1 + \alpha) v_1 - \alpha \times v_1 \times \dfrac{p_2}{p_1}} =$$

$$\frac{p_1 v_1 \ln \dfrac{v_1}{v_2}}{v_1} \times \frac{1 + \alpha - \alpha \times \dfrac{p_2}{p_1}}{1 + \alpha - \alpha \times \dfrac{p_2}{p_1}} =$$

$$= p_1 \ln \dfrac{v_1}{v_2} = p_1 \ln \dfrac{p_2}{p_1} = \dfrac{A}{V_{sg}} \text{ for } v_s = 0.$$

This is the general proof that the clearance volume has no influence on the energy consumed. The above mentioned means (the artificial enlargement of the clearance volume in the high pressure stage) may therefore be applied without hesitation to those machines in which, in contrast with the engines here considered, emphasis is laid on the energy requirement. The energy requirement for the compressor of the Diesel engine is somewhat less than 10% of the engine output and therefore plays no important role.

Instead of the above mentioned means for enlarging the high compression stage, a smaller r.p.m. may be chosen for the compressor. So long, however, as the r.p.m. of the engine does not exceed 1200, it is better, for the sake of simplicity, to drive the compressor directly from a crank of the crankshaft.

As to the purely structural form of the compressor, piping between the two stages and the receiver should be avoided on fixed engines. By suitable construction, all piping can be avoided, with the exception of the normal pressure intake pipe and the high pressure pipe. The air intake cock is located near the end of the normal pressure intake pipe, within reach of the driver's seat, and corresponds to the desired hand intake pressure (in case no automatic adjustment is provided). Small poppet valves may be employed as compressor valves, two or three for normal pressure and only one for high pressure. All poppet valves, including both suction and pressure valves, can be used interchangeably with each other and also for other purposes, for instance, as non-return valves, etc.

7. Starting device.- In most cases, compressed air is used for starting the engine. In order to maintain as large a supply of compressed air as possible, the flasks are charged up to 80 or 90 atm. and, by means of a pressure reducing valve, the starting pressure is reduced to about 20 atmospheres. The true starter consists, for example, of a device by which at first all the cylinders are connected with "air" and, after sufficient increase of the r.p.m. with the "air compressor intake." In order to increase the safety in starting, the cylinders may be switched, one after the other,

from "air" to "compressor," as is often done on marine engines. In this connection, it is assumed that each cylinder has its own intake valve. In case the starting valves are operated by compressed air (by means of so-called "starting-valves" and "distributing pistons"), no special modification of the engine is required, it being much more satisfactory to employ a starter, which should preferably be located within reach of the driver's seat. It will not then be necessary to leave the driver's seat for the purpose of cranking the engine. For stopping the engine, the starter is brought to the stopping position, at which the air intake is interrupted. At the same time, the fuel adjustment, which may be attached to the steering wheel in the same way as the gas throttle, must by placed on zero.

Instead of the arrangement described, a compressor may be provided, which works in starting, as a compressed air engine, or, in special cases, a small starting engine may be employed (explosion engine or electric starter).

In the case of a compressorless Diesel engine, we can dispense with the compressed air or any auxiliary device by giving the engine, with disconnected compression, the highest possible revolution speed by hand, and then by giving full compression in only one cylinder at first. This requires, however, considerable expenditure of energy, so that, in case there is no electric starter provided, the employment of a disconnectable compressed air starter is to be recommended, especially as the compressed air can also be used for the pneumatic tires.

8. <u>Fuel pump with valve gear and spraying nozzle</u>.- The fuel pump and the spraying nozzles are the most important parts of the Diesel engine. The experiments undertaken in this connection, and still more the suggestions made for the construction of these parts, are innumerable, notwithstanding that there is only one possibility for poly-cylinder engines, especially for swift-running engines. The fuel (however small in amount) required for each combustion must be measured to each cylinder by a special metering valve. Since the fuel pump cannot exert any real suction, the fuel must be led to it from the container under moderate compression of about 0.3 atmosphere, which can be generated by a small special air pump. In this connection, care must be taken to admit no air into the fuel pump and, since small air bubbles cannot be prevented from flowing with the fuel toward the pump, provision must be made for removing this air. Above all, the pump itself must be so constructed that not the smallest air bubble can remain in it.

Fuel pumps for poly-cylinder engines with only one metering valve and with so-called "distributors," for delivering the fuel in equal quantities to the different cylinders, have been abandoned as impracticable. Just as impracticable are the pumps in which, instead of measuring the fuel, the spraying pressure is regulated. For poly-cylinder engines, such regulating, which may indeed answer for a one-cylinder engine, always gives unequal fuel distribution, since the flow-resistance of the different nozzles is never the same, even with the most careful adjustment.

For intake valves (so-called fuel valves), needle valves with

strong springs have proven the best, with their cylindrical stems made tight with stuffing boxes.

If the spraying is done by means of compressed air, then the valve is forcibly opened (by a cam) during the spraying period and the air current injects the fuel which is located back of the valve, into the combustion chamber. The so-called open nozzles, through which only air is delivered (the fuel being delivered by means of a low pressure pump through an opening located between the air intake valve and the combustion chamber) are not so suitable for high revolution speeds, such as are here involved.

If the engine, on the contrary, is equipped with a mechanical fuel sprayer, then the needle valve does not need to be operated independently, but opens automatically under the pressure of the fuel pump and receives for this purpose a distributing (or metering) piston (similar to the needle valve of the English Ruston-Proktor four-stroke hot-bulb engine). Since the distributing piston cannot be kept perfectly air-tight and since leaks, however, would cause unequal distribution to the different cylinders, it is desirable to provide the distributing piston with an absolutely tight stuffing box.

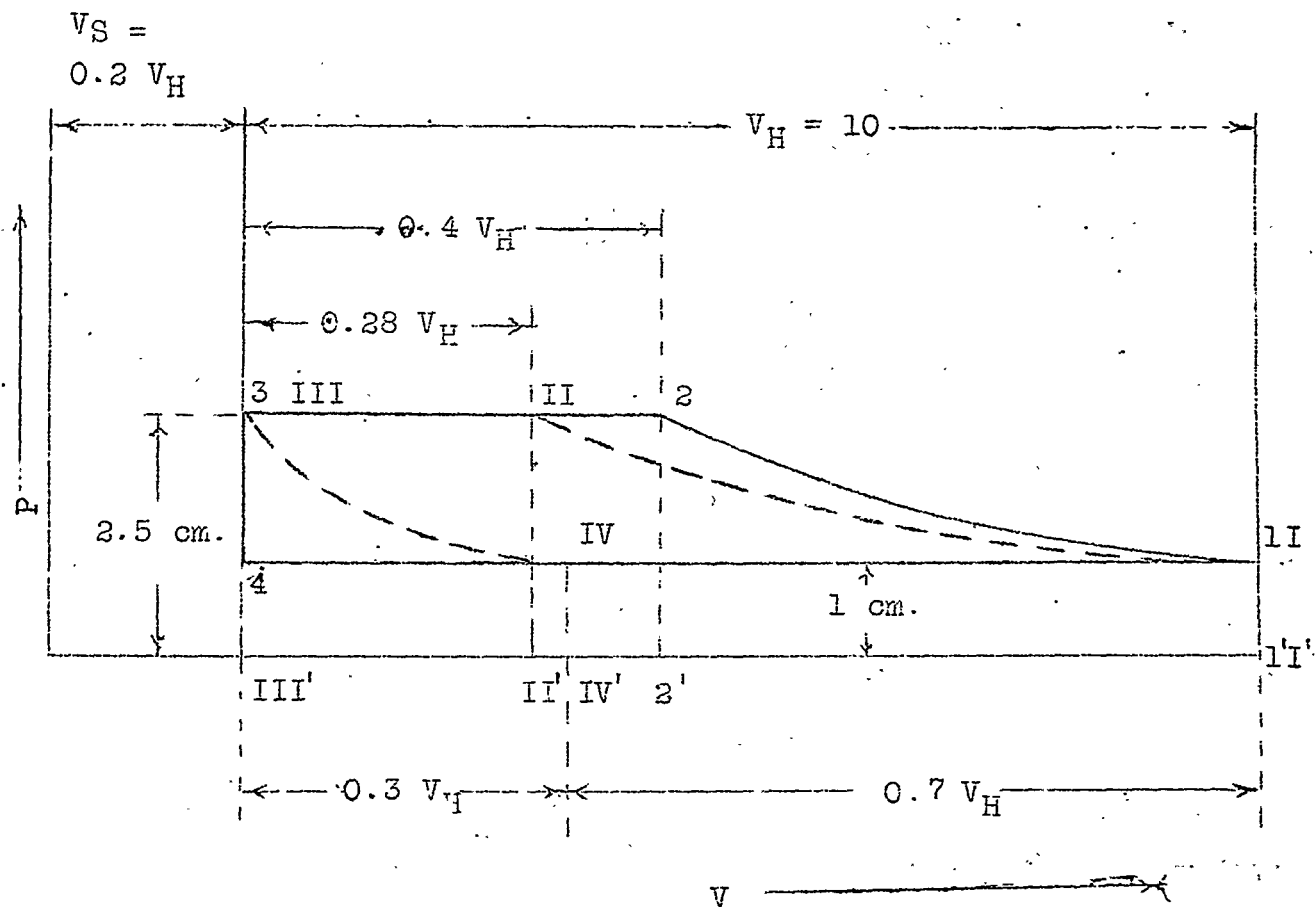(Translated from "der Motorwagen" January 20, 1920, by N. A. C. A.)

Fig. 1